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# Simulation model for the study of overhead rail current collector systems dynamics, focussed on the design of a new conductor rail

CARLOS VERA<sup>1</sup>, BERTA SUAREZ<sup>2,3</sup>, JENNY PAULIN<sup>2</sup> and PABLO RODRÍGUEZ<sup>2</sup>

## SUMMARY

Overhead rigid conductor arrangements for current collection for railway traction present some advantages in comparison to other, more conventional, energy supply systems. These advantages are simplicity, robustness and ease of maintenance, apart from its flexibility as to the required height for installation, which makes them particularly suitable for use in subway infrastructures. Nevertheless, due to the increasing speeds of new vehicles running on modern subway lines, a more efficient design is required for this kind of system.

In this paper, the authors present a dynamic analysis of overhead conductor rail systems, focussed on the design of a new conductor profile, with a dynamic behaviour superior to that of the system currently in use. This means that either an increase in running speed can be attained, which at present does not exceed 110 km/h, or an increase in the distance between the rigid catenary supports, with the ensuing saving in installation costs.

This study has been carried out using simulation techniques. The ANSYS programme has been used for the finite element modelling, and the SIMPACK programme for the elastic multibody systems analysis.

*Keywords: catenary, pantograph, overhead conductor rail, optimization, new design.*

## 1. OVERHEAD CONDUCTOR RAIL CURRENT COLLECTION FOR RAILWAY TRACTION

### 1.1. Introduction

In 1972, the French firm Delachaux [1] registered the first patent for an overhead rail conductor system, comprising an aluminium profile which clamped a copper wire, held in place by the elastic deflection of the aluminium rail (Fig. 1). However, this was not seriously implemented until 1983. In 1984, the Swiss firm Furrer+Frey [1, 2] adopted this system, and since then has considerably increased the total length of installations fitted with this type of electrification.

The main advantages of the overhead rigid conductor system are its simplicity, robustness and ease of maintenance. In addition, the fact that it is at a considerable height above the track avoids any risks of accidental contact, thus allowing for higher electric voltages to be used than with the traditional third rail. Another of its advantages,

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compared to other overhead collector systems, is its flexibility as to the required height for installation, which makes it particularly suitable for use in tunnels. It is this feature that has led to it being chosen increasingly for city underground systems as well as for main line tunnels and other underground railway lines.

This article presents the work undertaken by the Railway Technology Research Centre (CITEF – Centro de Investigación en Tecnologías Ferroviarias) of Madrid Technical University (UPM – Universidad Politécnica de Madrid), in collaboration with Metro de Madrid's Engineering Group. The aim of this project was to design a new rigid catenary profile, or overhead conductor rail, with a dynamic behaviour superior to that of the system currently in use. This means that either an increase in running speed can be attained, which at present does not exceed 110 km/h, or an increase in the distance between the rigid catenary supports, with the ensuing saving in installation costs. This study has been carried out using simulation techniques. In order to generate the catenary and pantograph models, the ANSYS programme was used for the finite element modelling and the SIMPACK programme for the multibody systems analysis.

## **1.2. Main features of current rigid catenary**

Rigid catenary [1-4] comprises a hollow aluminium profile rail, with an opening in the underside which acts as a pincer or pliers, thereby clamping the copper contact wire in place (Fig. 1). The aluminium rails come in lengths of between 5 and 12 m and are connected to each other by means of bridles, located and bolted on the inside of the profile, thus forming longer spans (up to 500 m), called overlap sections, whose total length is limited by the longitudinal thermal expansion of the rail.



Fig. 1. Conductor rail.

Each rail is suspended from the tunnel roof arch, its correct positioning being ensured by a support which allows for its vertical and lateral position to be regulated. These supports also allow for any longitudinal displacement that might occur as a result of thermal expansion. The distance between supports is kept constant throughout the length of each overlap section, except at the beginning of the section, where it gradually increases from 2 m up to its nominal length, which is equal to the length of the aluminium profile sections.

During the installation process, the supports (Fig. 2) are placed to either side of the centre of the track, giving the conductor rail a sinusoidal shape on the horizontal plane. This effect, known as 'stagger', spreads the wear uniformly across the width of the contact strip.



Fig. 2. Supports and conductor rail.

The separation zone between two adjoining sections is called an overlap section (Fig. 3). This presents a mechanical discontinuity that allows for the free expansion of both ends of the overlapping sections. This mechanical discontinuity is obtained by a lateral overlapping of both sections, with a separation of roughly 100 to 200 mm, which occurs over a length of approximately 3 to 6 m, with the ends of the rails being curved upwards to avoid any sudden jolts against the pantograph.

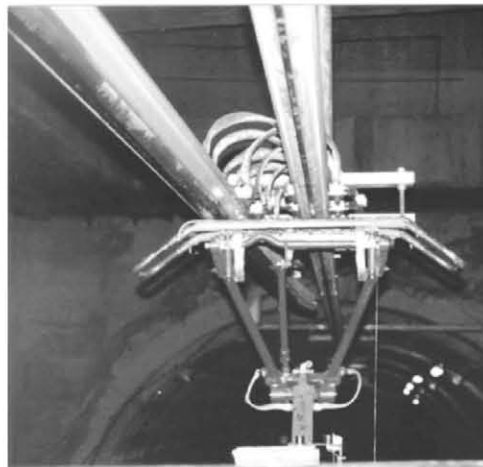


Fig. 3. Overlap section change.

## 2. CRITERIA FOR ASSESSING CURRENT COLLECTION QUALITY

Current collection quality depends on the dynamic behaviour of the system; the greater the uniformity of the catenary, the better it is. Also, the force exerted by the pantograph must be just great enough to prevent contact loss, and just small enough to limit wear. European railway administrations agreed to assess current collection quality by statistically processing the contact force between the catenary and the pantograph [5-8], while taking the following parameters into account:

- Mean value of the contact force,  $F_m$ ;
- Standard deviation of the contact force,  $\sigma$ ;
- Statistical minimum,  $(F_m - 3\sigma)$ ;
- Statistical maximum,  $(F_m + 3\sigma)$ , which must not exceed a certain safety limit;

- Number of contact losses lasting more than 0.01 s;
- Total accumulated contact loss time.

## 2.1. Standards

Among the various standards and guidelines for evaluating current collection quality, as well as the pantograph - catenary interaction, the following are worthy of mention:

- prEN 50318 [9]: Systematises the simulation programme input and output parameters, in order to be able to validate the pantograph – catenary interaction simulation methods.
- prEN 50367 [10]: Establishes the parameters needed to ensure interoperability, and provides the range of values within which the parameters of the system must fall. It recommends a higher limit for the mean value of the contact force according to speed: 120 N for 110 km/h and 135 N for 140 km/h. It also indicates the maximum standard deviation and the percentage of contact losses that might be reached, which for the cases studied are 0.3·F<sub>m</sub> and 0.2 %, respectively.
- UIC 794-1 [11]: Encourages the use of simulation programmes for analysing the interaction and provides a list of data needed to validate them. It includes recommendations on the mean contact force and the percentage of contact losses, which at 110 km/h must be less than 180 N and 0.14%, respectively.

## 3. DEVELOPMENT OF RIGID CATENARY MODELS

Bearing in mind its elastic behaviour, the catenary model was directly generated in ANSYS. The mechanical features of this model were then imported into SIMPACK, where the effect of the interaction with the pantograph was analysed, this latter being considered as a rigid multibody system with four degrees of freedom, as will be shown in the following paragraphs.

### 3.1. Modelling in ANSYS

For the catenary model to be set up using finite elements, it was structured into three groups:

- Conductor rail and contact wire: The conductor rail grips the contact wire with such force that all the forces are shared between both, which means that the wire contributes with its entire section to the rigidity of the set. This set has been modelled as a single body, for which the geometric properties (area, centroid, geometric moments) associated with an equivalent aluminium section, have been calculated.
- Conductor rail, contact wire and bridle: The geometric parameters for these sections are determined by adding the previous parameters to the ones for the bridle connector. However, it has been proved that the bridles have practically no effect on the system's dynamic behaviour, and therefore need not be taken into account.
- Elastic supports: An experimental test was made to adjust the vertical stiffness of a support that elastically holds the conductor rail and allows for longitudinal displacement of the rail.

The influence of certain parameters on the system behaviour was analysed by means of a comparative study. To this end, a reference model was defined, based on which a series of variations was carried out, which will be described later. In order to avoid excessively long calculation times during the initial stages of the study, this model corresponds to a straight span of only 57 m, and incorporates double elastic supports whose distance gradually increases from 2 to 10 m. Euler-Bernoulli unidimensional beam elements were used to model the rail, and spring elements for the supports. For the final calculations, 300 m long models were used, corresponding to an overlap section.

Once the model had been generated, the first eigenfrequencies and the eigenmodes were calculated.

### **3.2. Introducing the catenary as an elastic body into SIMPACK**

Having completed the modal analysis of the catenary, the results given by ANSYS were exported to SIMPACK, in order to allow the modelling of its interaction with the pantograph, as will be described in the next paragraphs. To be exact, the mass and stiffness matrices were imported into SIMPACK, as part of the equations defining the dynamic behaviour of the system as a whole. The nodal displacements of the elastic body were also defined by superimposing the eigenmodes, whose amplitudes had to be calculated with respect to time, when solving the equations defining the dynamic behaviour of the whole system.

Next, the static equilibrium under the influence of gravity was calculated, this new position being saved as the initial deformation of the elastic body.

Other characteristics peculiar to a rigid catenary have also been modelled in SIMPACK, such as damping, stagger from the track centre line (Fig. 4) and change of overlap section.

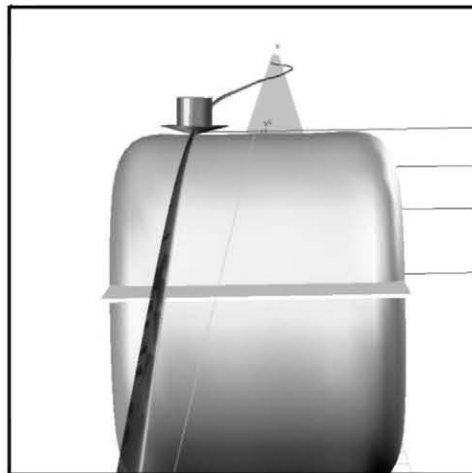


Fig. 4. Catenary model, showing stagger from the track centre line.

Damping was determined experimentally, and was found to be negligible. It was included as natural damping applied to each of the eigenmodes.

In order to model stagger, a series of points or markers were defined in the inertial system, which are located in the definitive position of the supports. Between these markers and equivalent markers on the same longitudinal position on the elastic body, high stiffness springs are placed. After calculating the static equilibrium, the springs position the catenary markers over the equivalent points of the inertial system, the elastic body being deformed to adapt to the required stagger. Likewise, the upward



curvature of the rail at the end of an overlap section is defined following this same technique.

At this point, the pantograph can be set up in SIMPACK to form a model of the catenary – pantograph set, and the appropriate simulations can be carried out.

#### 4. DEVELOPMENT OF PANTOGRAPH MODELS

In the dynamic studies of the pantograph – catenary interaction, generally focussed on the elastic behaviour of the latter, the use of simple pantograph models is fairly usual [5, 12, 13]. The fine-tuning of the overall model was undertaken with this kind of simple pantograph, defined from data supplied by the manufacturers. However, when the results obtained were analysed, some unexpected tendencies were found. Therefore, it was decided to build a more accurate pantograph model ourselves.

Since a simulation tool was available for modelling the catenary, which admitted elastic bodies for modelling the catenary, a three-dimensional pantograph model (Fig. 5) was chosen, that would include all the features of the real mechanism [14]. Figure 5 also shows a functional diagram of the pantograph model used [15].

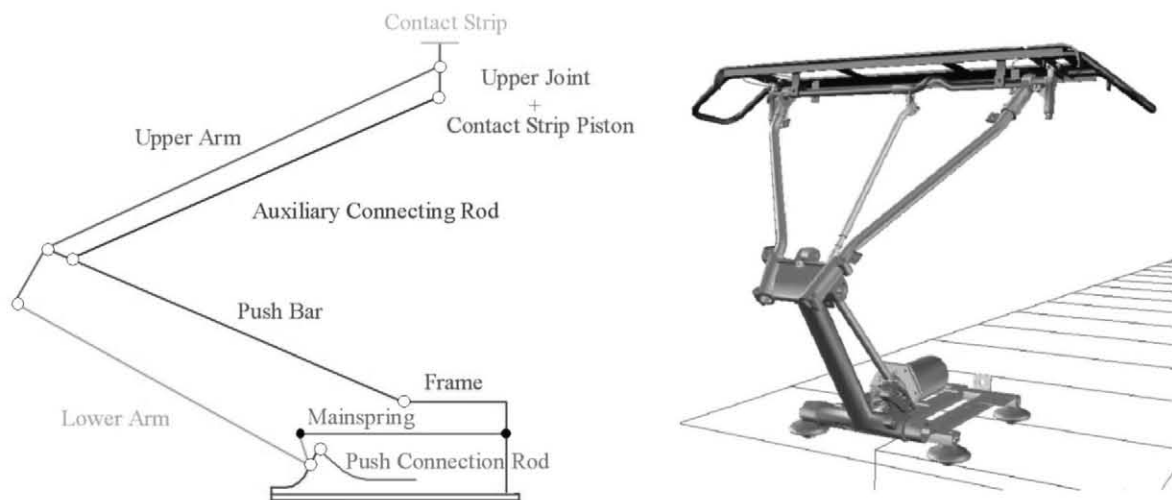


Fig. 5. Pantograph mechanism diagram and complete model.

These models are generated by defining a set of rigid bodies linked to one another by kinematic joints and force elements (springs, dampers, etc.).

A main spring was also included to ensure the application of a static force, and two vertical springs for the suspension of the pantograph head, thereby enabling it to adapt to the tiny irregularities of the catenary. Energy loss by friction in the joints was also taken into account and these were adjusted experimentally.

The mechanism thus defined presents four degrees of freedom of movement: raising of the pantograph mechanism and bouncing, rolling and pitching movements of the pantograph head (Fig. 6).

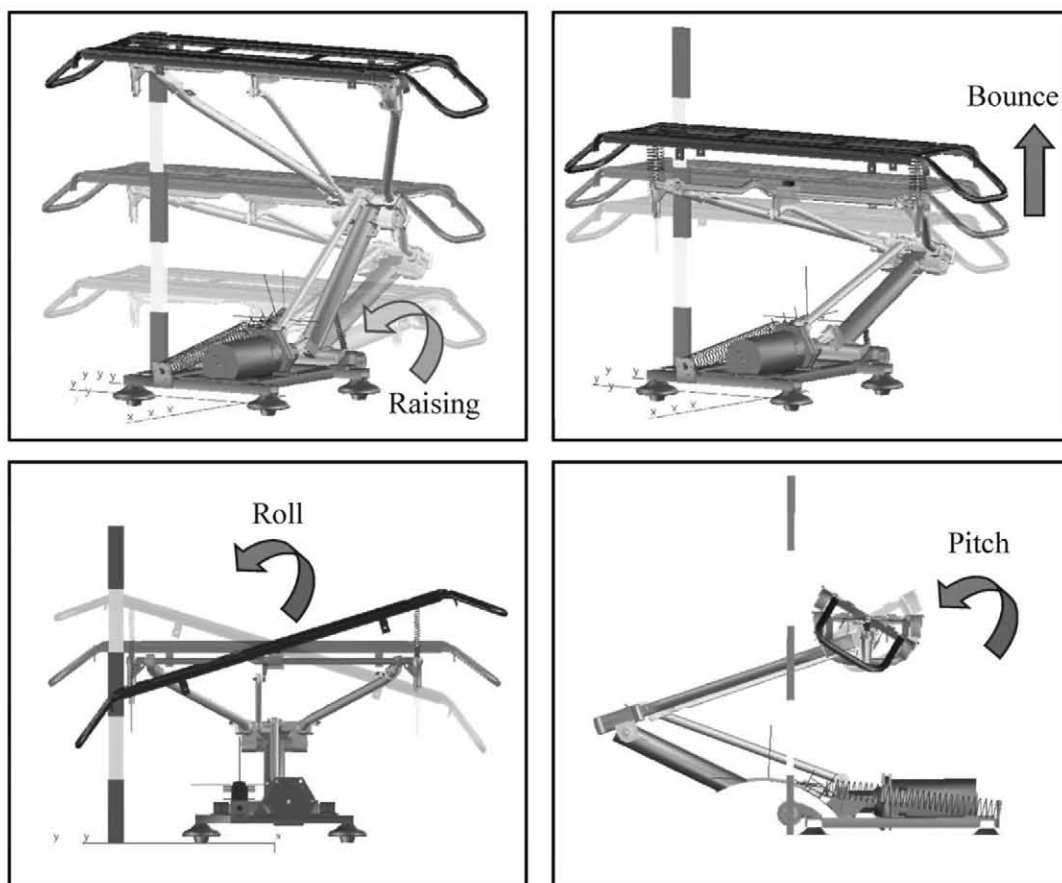


Fig. 6. Pantograph degrees of freedom: raising of the mechanism (up-left), bounce (up-right), roll (down-left) and pitch (down-right).

Also taken into account were the pantograph raising mechanism, the aerodynamic force, which at 110 km/h is 10 N (Fig. 7), and the bump stops that limit the pitching movement of the head, for angles greater than 5°.

### Pantograph aerodynamic force

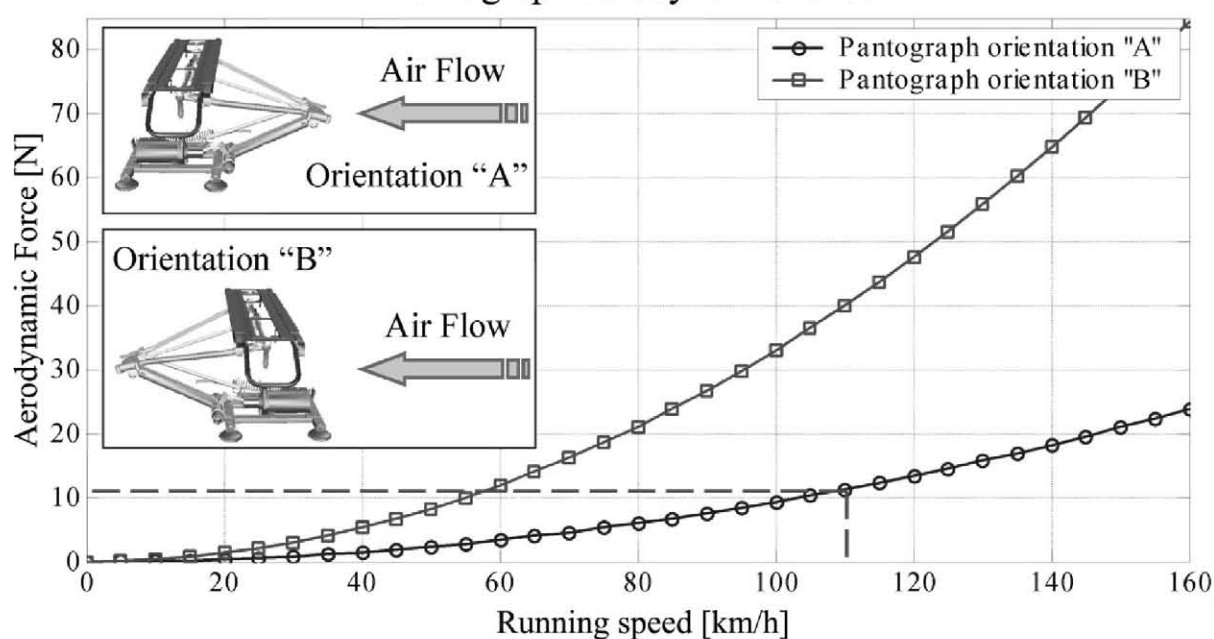




Fig. 7. Pantograph aerodynamic force: orientation 'A', with mechanism vertex directed towards the running direction, and 'B' in the opposite direction.

## 5. DEVELOPMENT OF THE INTERACTION MODELS BETWEEN PANTOGRAPH AND CATENARY

Having defined a model for the pantograph and the catenary, a static equilibrium calculation was made in order to adjust the operating height of the pantograph, which moves longitudinally at constant speed.

### 5.1. Contact model

Basically, there are two possible options when it comes to dealing with the problem of contact between the two bodies:

- For the relative position of both bodies to be of importance, but not their geometry (simple contact);
- For the geometry to be relevant in the contact problem ("complex" contact model).

In the simple model, the contact depends solely on the relative position between two points or markers located on the contours of each body. The contact is solved by connecting both points by means of a special spring that only comes into play when the distance changes sign, and so avoids the bodies penetrating each other.

In the complex model, the minimal distance between the surfaces that form the geometry of the bodies is calculated, and a spring, equal to the previous one, is defined between the moving points that form this distance.

The complex contact model has been used in this study because of the following advantages:

- The point of contact is positioned freely on the surfaces of the wire and the pantograph head;
- The pitching movement of each contact strip can be taken into account;
- A more uniform result is obtained during the integration of the movement equations.

A large radius closed cylindrical surface (Fig. 8) was used for the pantograph strips in order to get as near as possible to the flat shape of the strips. Two points of contact were used, one located on the front strip and another one located on the rear strip, in order to take account of the pitching movement of the pantograph head.

A small sphere or cylinder of the same diameter as the wire (Fig. 8) was defined for the catenary. This runs the length of the catenary, always positioning itself in the same longitudinal position as the pantograph.

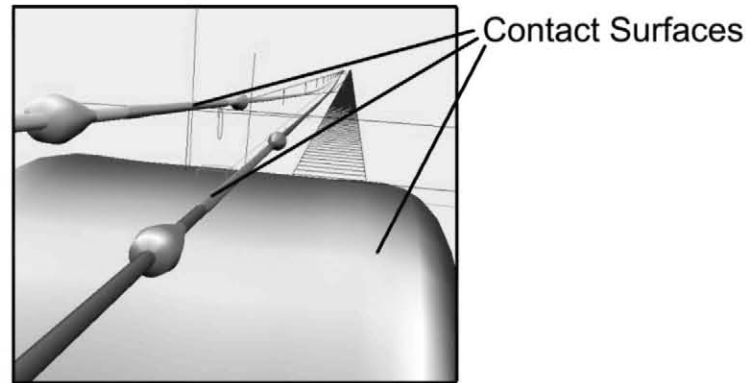


Fig. 8. Surfaces for the catenary and the pantograph.

## 5.2. Other features of the model

### 5.2.1. Moving marker for the overlap section change

This marker, which is used to define the contact point, is initially fixed at the beginning of the overlap section, but does not follow the pantograph until this latter has reached this position. The marker then follows the longitudinal movement of the pantograph as it runs along the overlap section, and comes to a stop on reaching its end.

### 5.2.2. Influence of contact spring stiffness

It is extremely important to ascribe a suitable value to the stiffness of this contact spring. Very high values give a “stiff” behaviour for the movement equations, particularly for high running speeds. On the other hand, an excessively weak contact gives rise to somewhat unrealistic results.

In the models presented in this work, a value of 50 000 N/m has been taken for the stiffness of each contact spring (two for each pantograph head), in accordance with what is set down in regulation prEN 50318.

### 5.2.3. Influence of the number of rigid catenary eigenmodes included in SIMPACK

Although a large number of eigenmodes gives more exact results, this is limited by the capacity of the computer and thus the computation time needed for the simulation. After a comparative study, it was found that computation time could be considerably reduced, with no noticeable negative effects on the precision of the results, by including the eigenmodes associated with frequencies of up to 30 Hz, or 20Hz if only vertical eigenmodes were of interest.

### 5.2.4. Influence of the distance between catenary nodes

As is the case with the modes, precision increases with the number of nodes used in the finite element model, but with a considerable increase in calculation time. This means that a compromise solution between precision and computational time had to be reached.

After a comparative study, it was concluded that, in the model adjustment stage, an inter-nodal distance of 1 m could be adopted, although, for the final calculations, it is preferable not to exceed 0.5 m.

### 5.3. Statistical analysis of the contact force

When the models have been set up and adjusted in SIMPACK, the next step is to solve the differential equations that represent the dynamic behaviour of the system, thereby obtaining, among other results, the contact force acting between the strips and the catenary. This result is then statistically processed in the way recommended by European railway administrations for the application of criteria for assessing current collection quality.

## 6. INFLUENCE OF THE DIFFERENT PARAMETERS CHARACTERISING THE SYSTEM

With the aim of analysing the main parameters defining the dynamic behaviour of the pantograph – catenary system, a series of comparative studies were undertaken based on the reference model, with each of the parameters being modified one by one. The parameters taken into account were: the geometric moment of area of the conductor rail cross-section,  $I_{VER}$ , calculated with respect to the vertical axis that passes through its geometric centre, the moment of area with respect to the horizontal axis,  $I_{HOR}$ , Young's modulus,  $E$ , the material density,  $\rho$ , the stiffness of the supports,  $k$ , the width of the supports,  $D$  and the damping of the catenary,  $d$ . In order to study their influence, the values of these parameters were doubled independently of each other.

Figure 9 and Figure 10 show the contact forces obtained in the reference model and by doubling the moment of area with respect to the horizontal axis.

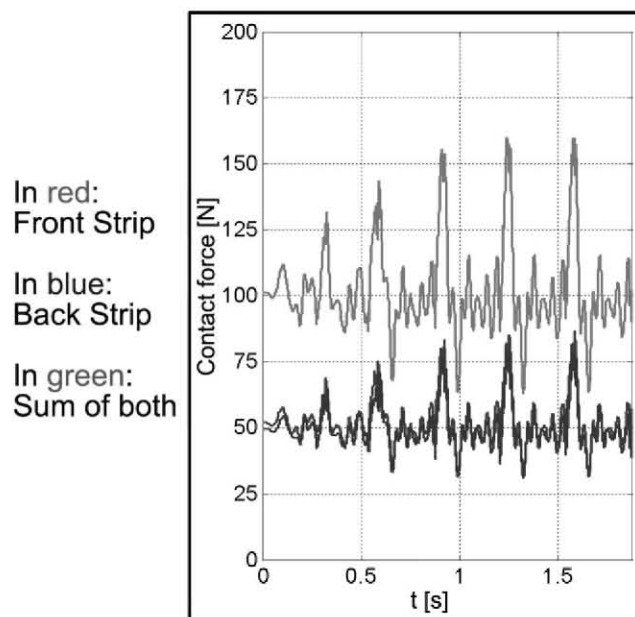


Fig. 9. Contact force for the reference model.

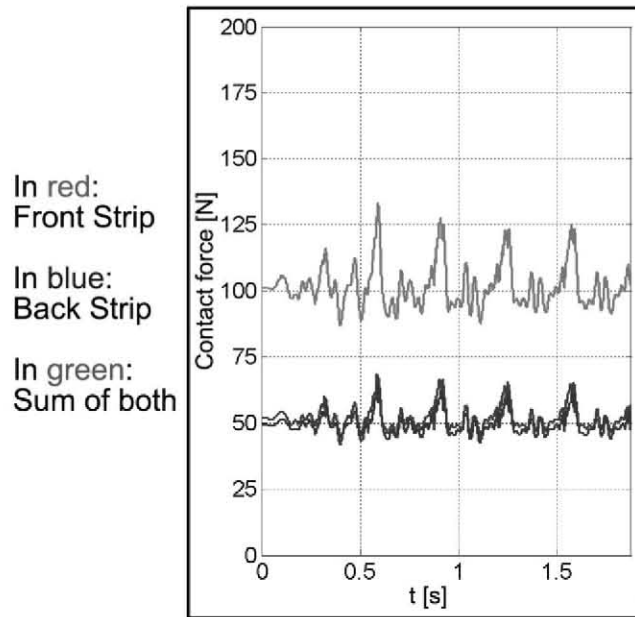


Fig. 10. Contact force when varying the moment of area  $I_{HOR}$ .  
It is desirable for the standard deviation to be small, since a homogeneous contact force favours the current collection process.

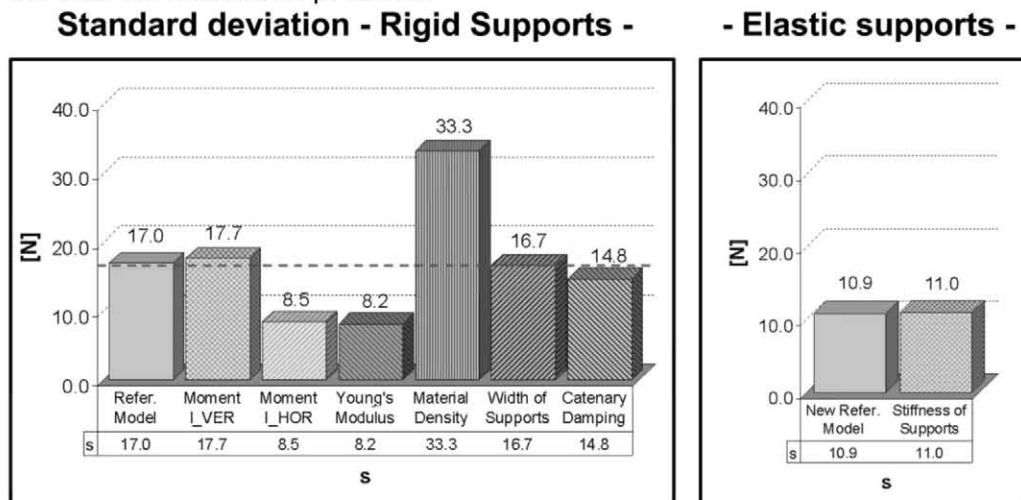


Fig. 11. Standard deviation, for rigid supports (left) and elastic ones (right).

As can be seen in Figure 11, when the horizontal moment of area,  $I_{HOR}$ , is doubled, the standard deviation is reduced to one half. The same effect is produced by doubling Young's modulus,  $E$ . On the contrary, when the density is doubled, the standard deviation is also doubled.

According to the previous results, it would seem advisable to increase the geometric moment of area,  $I_{HOR}$ , and Young's modulus, while an increase in density worsens the dynamic behaviour. With regard to the moment of area with respect to the vertical axis of the transversal cross section,  $I_{VER}$ , the stiffness of the supports,  $k$ , the damping of the catenary,  $d$ , and the width of the supports,  $D$ , it was seen that their variation had hardly any influence on the dynamic behaviour of the whole system.

The purpose of this study being the design of a new catenary profile, no pantograph parameters were taken into account for this sensitivity analysis.

## 7. DESIGN OF A NEW RIGID CATENARY PROFILE

### 7.1. Prior considerations

The design process was focussed on improving the dynamic behaviour of the catenary – pantograph system for running speeds equal to, or higher than 110 km/h. This improvement is aimed either at increasing the separation of supports to a distance greater than at present (10 m) or at increasing the actual maximum running speed (110 km/h), should the distance between supports not be changed.

Also, it was decided to maintain the original conductor rail design, whereby the rigid catenary would comprise a copper contact wire, held in place by an aluminium profile, which meant that a system for fastening both bodies together needed to be considered. All the many other solutions considered for fastening the contact wire to the rail needed the incorporation of intermediate elements, such as bolts, clips, etc., which would complicate the installation procedure. As the only known solution for fastening the copper wire without using external elements is the one used in the present design, which takes advantage of the aluminium's elasticity, it was decided to maintain the lower part of the profile design, that allows clipping of the copper wire by means of the elastic deformation of the aluminium rail.

Given the results of the initial study, it was decided to increase the moment of area,  $I_{HOR}$ . Two additional design factors were also taken into account:

- As described above, the aluminium profile has two flanges, or clamping arms, which retain the contact wire through elastic deflection prestressing. The pressure applied by the aluminium rail to the copper wire must be high enough to grip the wire firmly, but not too high to cause any plastic deformation.
- The force required to splay the rail during the assembly process should not be very great in order not to cause any plastic deformation of the material.

### 7.2. New design

Of all the different designs taken into consideration for the rail, the profile shown in Figure 12 was deemed the most appropriate:

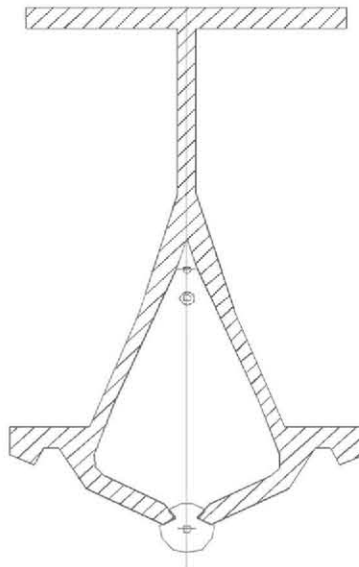


Fig. 12. New Design: METRO-730 profile.

This new profile is 150 mm high (compared to 110 for the original profile), which allows for a better distribution of the mass without causing any problems of incursion into the tunnel clearance.

## 8. GEOMETRIC COMPARISON OF THE NEW AND OLD PROFILE

Table 1 shows the percentage increases in the geometric characteristics of the new design.

Property	Increase in the new profile in respect of the present one
Equivalent area	- 1.24%
Equivalent density	+ 0.14%
Moment of area $I_{VER}$	- 4.25%
Moment of area $I_{HOR}$	+72.04%

Table 1. Percentage increases in the mechanical properties of the new profile section.

As can be seen, a substantial increase in the horizontal moment of area  $I_{HOR}$  has been obtained, while the other parameters, according to established design criteria, have scarcely suffered any alteration.

## 9. 3D MODEL STATIC BEHAVIOUR COMPARISON

The behaviour of both profiles was compared during the opening process by the conductor clipping carriage. This carriage has two pairs of wheels that fit into grooves in the conductor rail (Fig. 13). When each pair of wheels is separated, pressure is applied to the rail causing the pincer to open, so that the wire can be inserted.

In order to analyse the opening process, three dimensional models are needed. A 90 cm long segment is sufficient to obtain valid results, as the deformations caused by the carriage spread over a 60 cm zone. This segment is considered as a clamped-clamped beam, so as to represent the effect exerted by the rest of the rail on the section under study and to prevent any relative displacement or rotation between both.

Although the opening forces possess both a vertical and horizontal component, the former can be annulled by the reaction of a second set of wheels resting on the horizontal plane located above the pincer (Fig. 13). Therefore, the carriage force was defined by using horizontal point forces applied to the ends of the pincer, shown in Fig. 13.



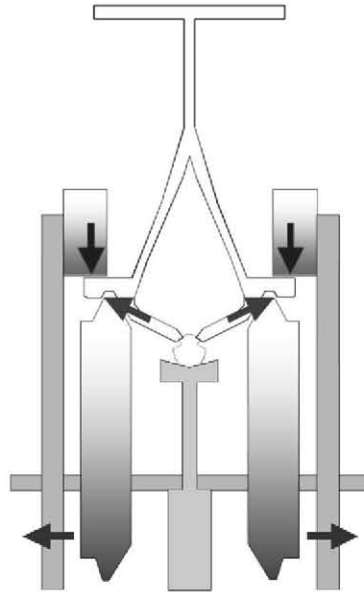


Fig. 13. Outline of how the present clipping carriage works.

Figure 14 shows how the stress is distributed. The blue contours indicate the lowest stresses, and the red ones, the greatest. As can be seen in the figure, in the centre part, the stresses in the new design are higher than in the original model, and are no longer to be found in the top part of the profile flanges, but stretch throughout the length of the shorter flanges of the new design. However, the highest stress values appear in the load application section.

The greatest stress appears just at the points where the forces are applied, that is, at the ends of the pincer. In both cases, the elastic limit is slightly exceeded. However, the simplification chosen to model the loads means that the stress will be somewhat less in the real situation.



Fig. 14. Von Mises stress in the centre part, in the present model (left) and in the new one (right).

To prevent plastic deformations occurring in the new profile, and to improve the assembly process, a new design of carriage with four pairs of wheels was chosen in order to distribute the loads better and reduce the maximum stress. When this case was

modelled, it was seen that the greatest stress was considerably lower as a result of distributing the total force over eight wheels instead of four.

The results are gathered in Table 2, so that they may be more easily compared:

Profile	No. loads	Unitary Force	Total force	Stress eq. (centre)	Stress eq. (application F)
					Inner / Outer wheel
<u>present</u>	4	750 N	3000 N	39 MPa	161 MPa
new	4	925 N	3700 N	54 MPa	172 MPa
	8	472 N	3776 N	52 MPa	91 / 89 MPa

Table 2. Comparative results.

## 10. COMPARATIVE STUDY OF THE DYNAMIC BEHAVIOUR OF PROFILES

To compare the dynamic behaviour of the new profile with the present one, three different simulations were undertaken in SIMPACK. The cases studied were as follows:

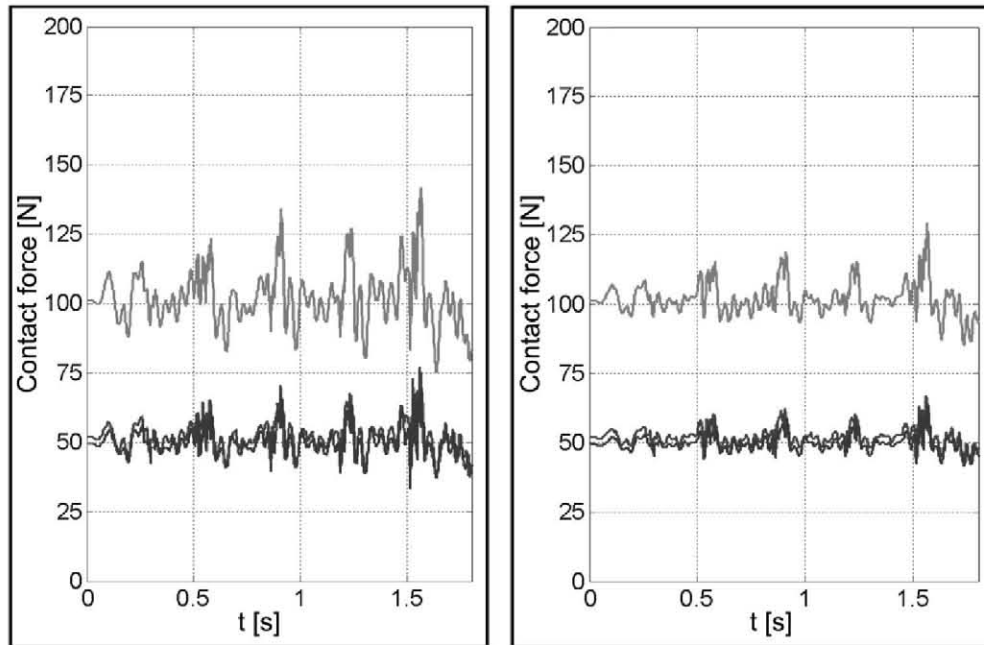
- Comparison between the original profile and the new design;
- Determination of the maximum distance between supports;
- Determination of the maximum speed attainable.

### 10.1. Comparison between the original profile and the new design

A direct comparison was made between both profiles, with their dynamic behaviour being simulated for a 10 m distance between supports, and a 110 km/h running speed.

Figure 15 shows the contact force obtained in each case. As can be seen, the curves corresponding to the contact force of the new profile are smoother and oscillate less than those of the original profile.

**Current Profile, 10 m, 110 Km/h      New Profile, 10m, 110 km/h**



In red: Front Strip; In blue: Back Strip; In green: Sum of both

Fig. 15. Contact forces for a double-strip pantograph.

Figure 16 shows the statistical results. It can be seen that the standard deviation is reduced to almost half, which shows that the dynamic behaviour of the new design is better.

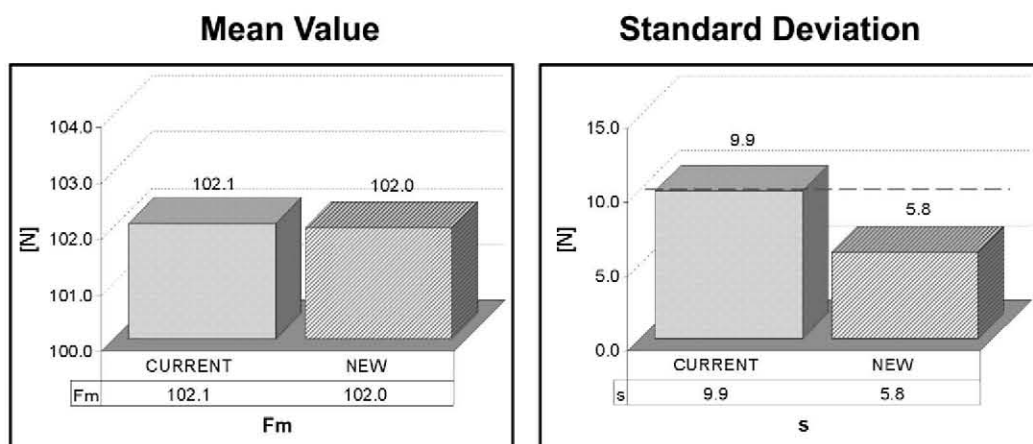
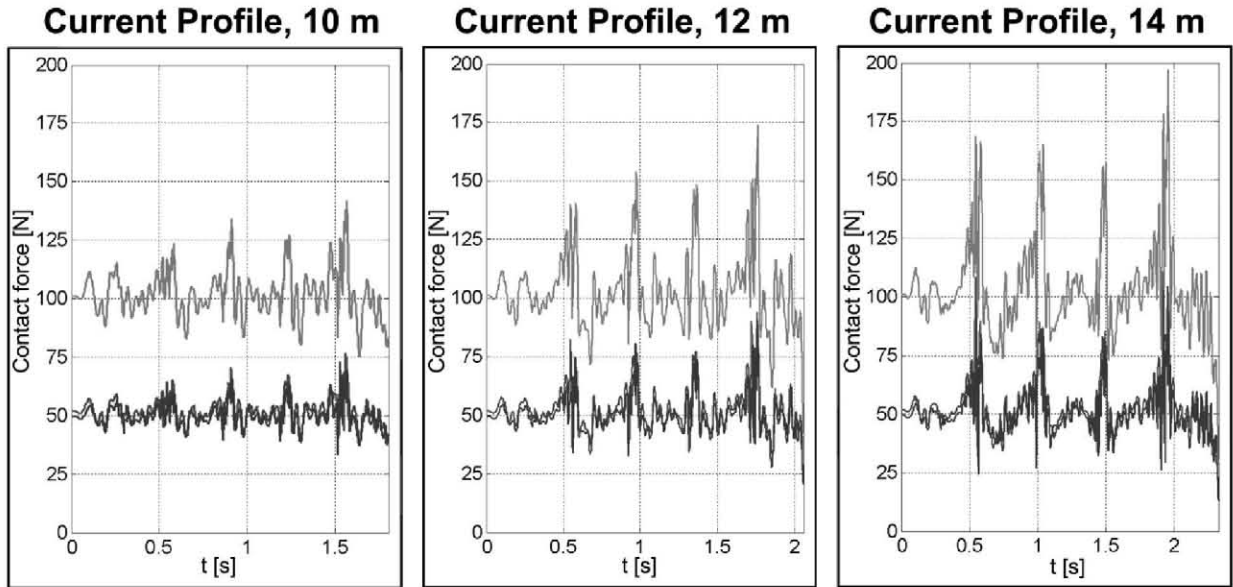


Fig. 16. Contact force statistics for the current and new profiles.

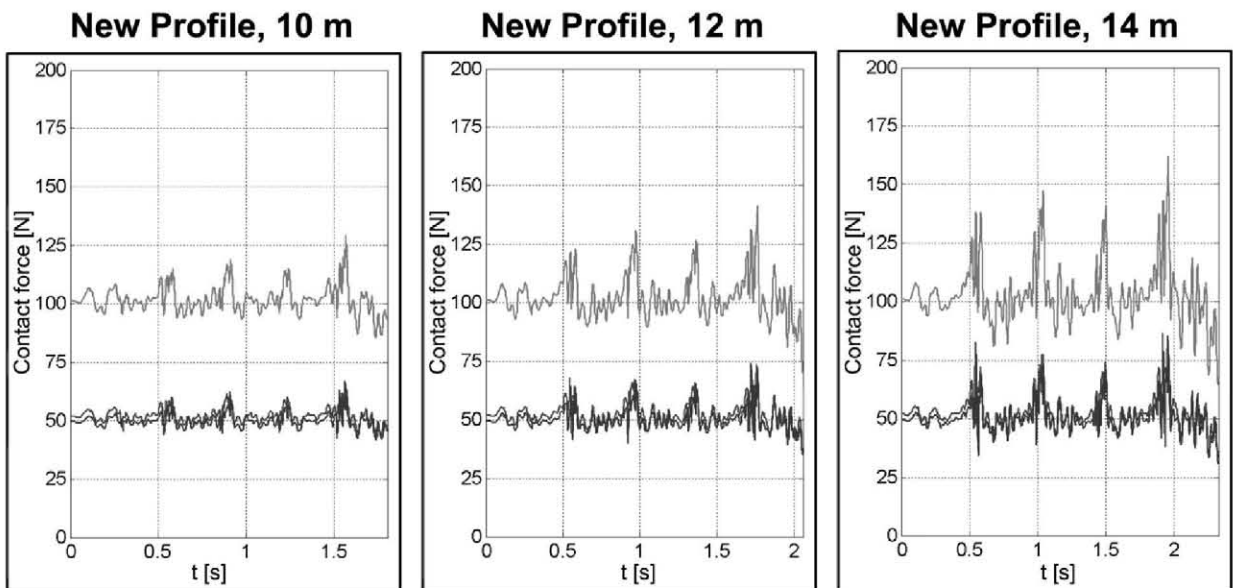
## 10.2. Determination of the maximum distance between supports

It was also decided to discover to what extent the distance between supports in the new design could be increased, so as to obtain a dynamic behaviour similar to the present one. Figures 17 and 18 show the contact force obtained for the present conductor rail and for the new one, with a distance between supports of 10, 12 and 14 m.



In red: Front Strip; In blue: Back Strip; In green: Sum of both

Fig. 17. Simulated contact forces for the current profile and a double-strip pantograph.



In red: Front Strip; In blue: Back Strip; In green: Sum of both

Fig. 18. Simulated contact forces for the new profile and a double-strip pantograph.

It can be seen (Figs. 17 and 18) that, with the new profile (12 m), a very similar dynamic behaviour to that of the reference case (current profile, 10m) is obtained. However, with the new profile and supports spaced at 14 m intervals, a slight worsening of the behaviour can be observed, compared to the current rail supported every 10 m.

Figure 19 shows the statistical results for the contact force.

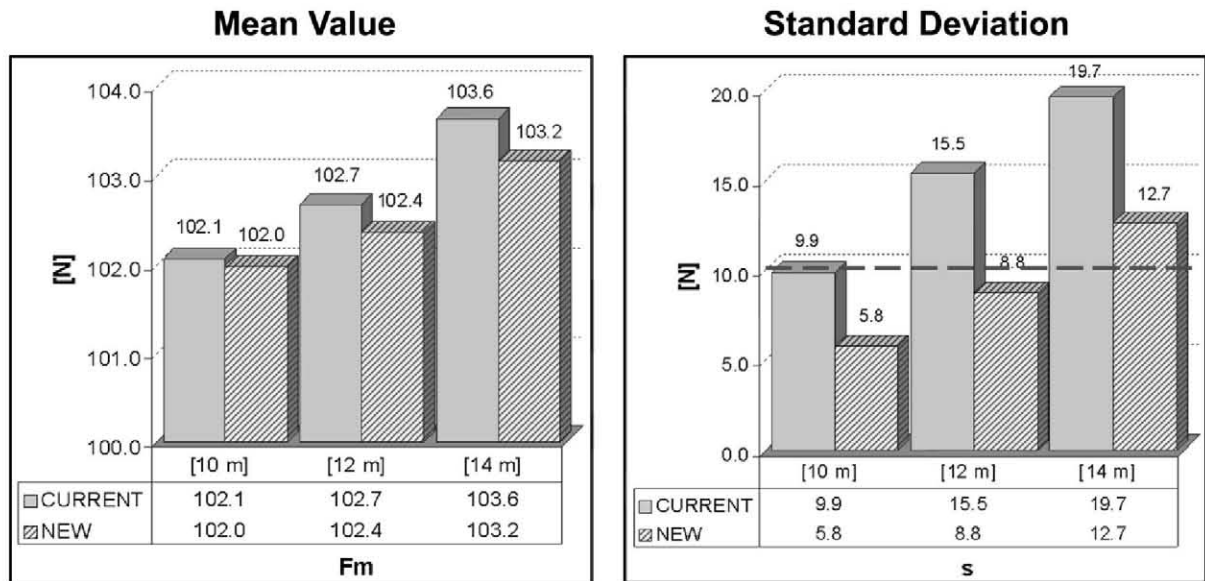
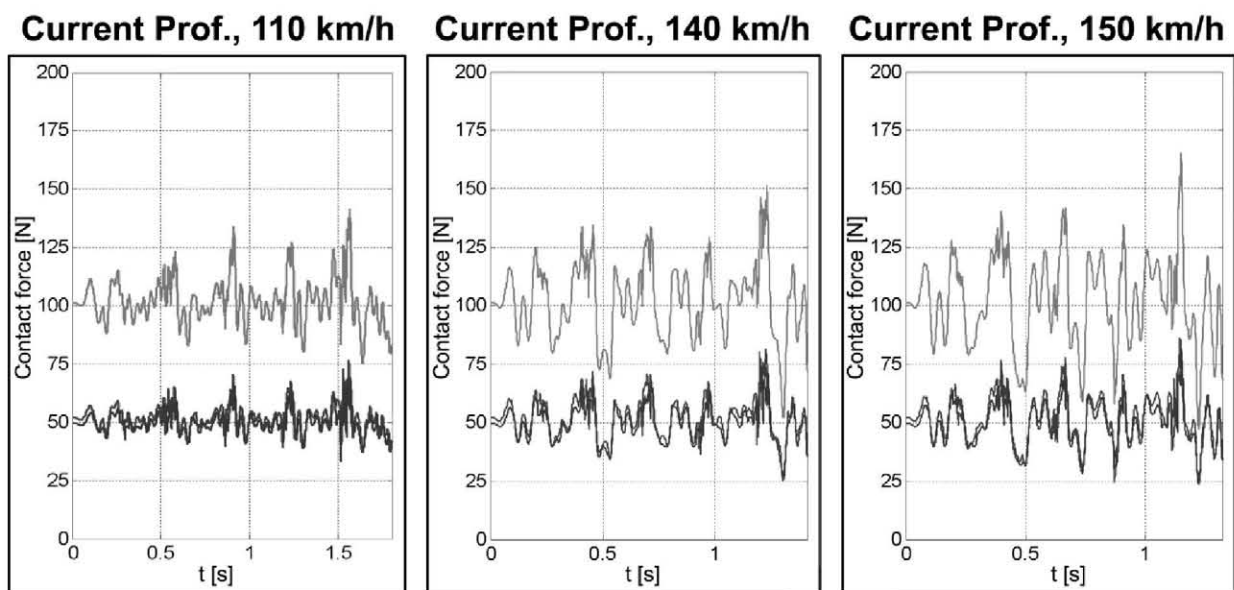


Fig. 19. Contact force statistics, for the current and new profiles.

The standard deviation for the new profile with a distance between supports of 12 m is very similar to the one corresponding to the present situation (current profile, 10 m). A slightly worse dynamic behaviour is shown for the new profile, when increasing the separation of supports from 12 to 14 m.

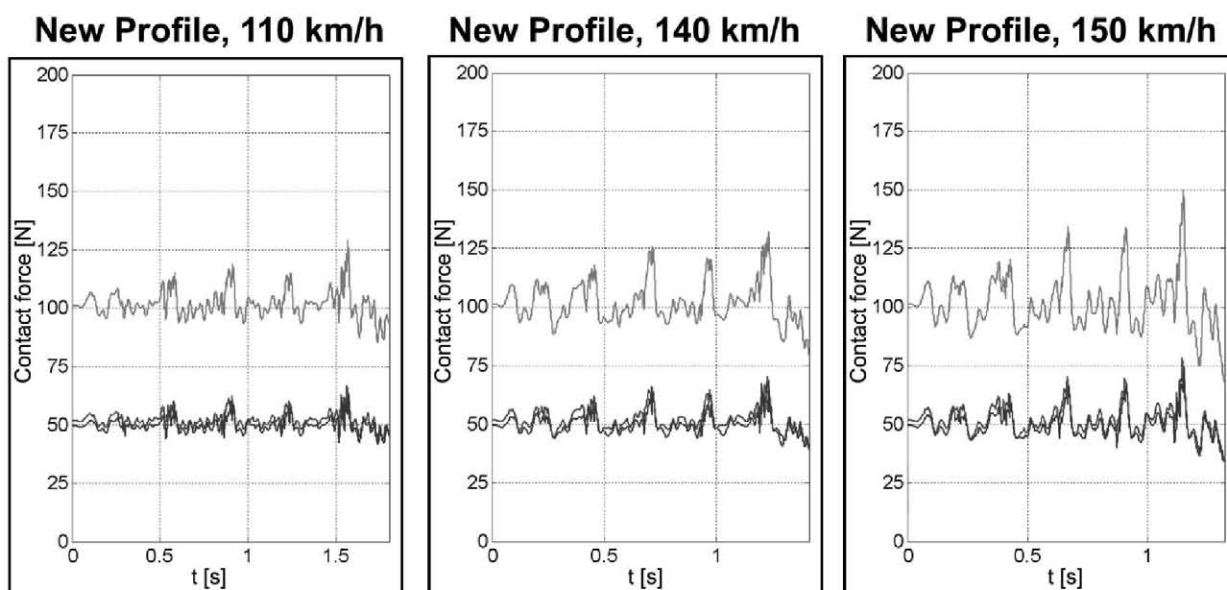
### 10.3. Determination of the maximum speed attainable

Further simulations were aimed at finding the maximum speed that could be reached with the new profile while obtaining a similar dynamic behaviour to the present one. Figures 20 and 21 contrast the contact force curves obtained for the original configuration with those found for the new rail at running speeds of 110, 140 and 150 km/h.



In red: Front Strip; In blue: Back Strip; In green: Sum of both

Fig. 20. Contact forces for the current profile and a double-strip pantograph.



In red: Front Strip; In blue: Back Strip; In green: Sum of both

Fig. 21. Contact forces for the new profile and a double-strip pantograph.

The new profile configurations at 140 and 150 km/h show a fairly similar dynamic behaviour to the reference case (current profile, 110 km/h), although the graphs relating to the new profile are smoother.

Figure 22 shows the statistical results for the original and new profiles. As can be seen, the statistics for the present situation (current profile, 110 km/h) fall between the ones relating to the new profile at 140 and 150 km/h.

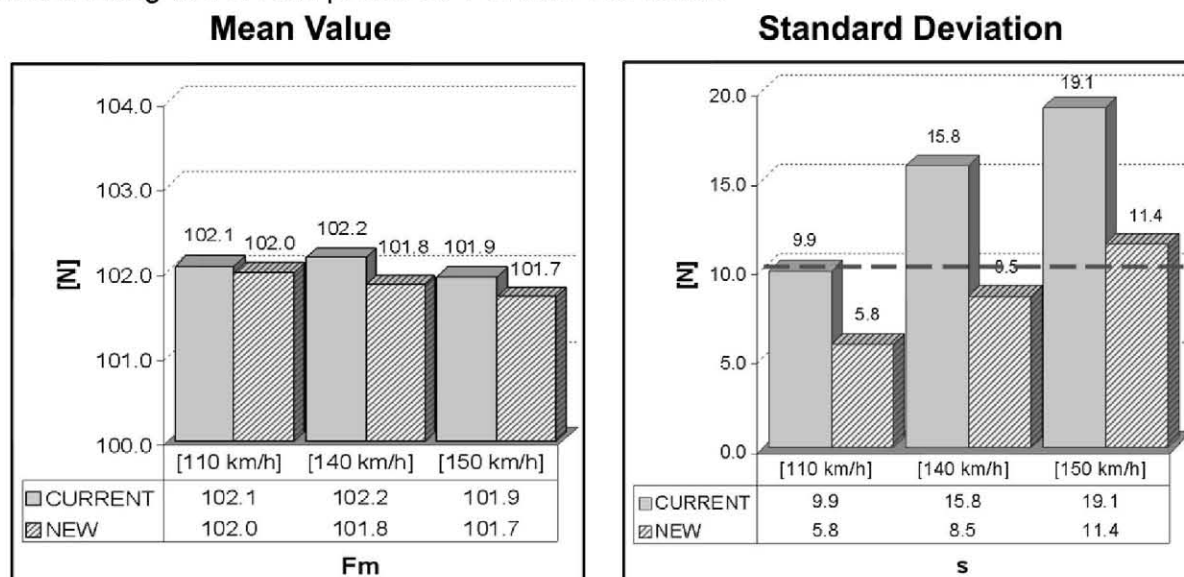


Fig. 22. Contact force statistics, for the current and new profiles.

## 11. TESTING STAGE

Considering the excellent results found in the previous simulations, Metro de Madrid and CITEF applied for a formal patent for the new designed conductor rail. This patent [16] was officially accepted in December 2004 (European patent nº EP 1 484 214 A1).



Metro de Madrid also requested the construction of a prototype (Fig. 23). This prototype was recently installed in a test track sited in their own network, comprising a total length of 500 m.



Fig. 23. New conductor rail prototype.

At the present time, the new overhead conductor rail is being tested. The correct performance of the whole system will be assessed under real service conditions. A field measuring process will be conducted, in which the data acquisition will be realised by a measuring pantograph developed by CITEF (Fig. 24).



Fig. 24. Measuring pantograph

The measuring pantograph is equipped with load cells under both springs of the strip suspension, and with two accelerometers for recording the vertical movements of the pantograph head. By properly combining these signals, an indirect measurement of the contact force can be achieved.

Although no definite experimental results are available yet, comparisons of first results reveal the superiority of the new design as was predicted in simulation. In a future publication, the authors will present the experimental results obtained.

## 12. CONCLUSIONS

It has been shown, by means of simulations, that this new conductor rail design for an overhead third rail current collection system leads to an improved dynamic behaviour compared to the rail currently in use. It is possible to attain higher running speeds without reducing the distance between supports. Moreover, on the lines where there is

no wish to exceed running speeds of 110 km/h, a new profile rail can be installed with slightly greater distances between supports, resulting in a saving in materials and installation time.

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